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# Organic Rankine cycles in waste heat recovery: a comparative study

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## Abstract

A theoretical study of organic Rankine cycles (ORCs) powered by three different waste heat sources is presented. The heat sources, all found in industrial processes, span a range of energy scales capable of powering ORCs from  $\sim 10$  kW to 10 MW. A novel method of pinch point analysis is presented, allowing variable heat input to the ORC. This study models the ORC over a range of operating conditions and with different working fluids for each heat source. Results from each source are compared to assess the influence of different heat source characteristics on optimal ORC design.

**Keywords:** organic Rankine cycles; waste heat recovery

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## 1 INTRODUCTION

Waste heat is an inherent and abundant byproduct of many industrial processes. Often, industrial waste heat is too low in temperature for any useful energy to be extracted from it by conventional cycles (e.g. steam turbines). Organic Rankine cycles (ORCs) rely on the same principle as conventional steam/water Rankine cycles used for primary thermal power generation purposes. ORCs use organic working fluids. These fluids have lower boiling points than steam/water at the same pressure, which allow them to be driven by low-grade waste heat. ORCs functioning down to  $73.3^{\circ}\text{C}$  have been reported [1]. By converting waste heat into mechanical power, ORCs can improve the efficiency of a wide range of thermal cycles.

Several studies model ORC cycles theoretically [1–5]. Roy *et al.* [2] investigate the optimal ORC configuration and working fluid for an ORC system powered by waste flue gas from a  $4 \times 20$  MW power station. However, the optimization is constrained by using a constant heat supply. This approach may be appropriate for flue gas, as there is a limit to the cooling of flue gas to prevent corrosion of components. It may not be appropriate to extend this model to other applications where there is no lower limit on the source heat temperature. Other studies use commercial software to model ORCs [3, 4]. Aneke *et al.* [3] use IPSEpro to simulate the ORC cycle of the Chena plant. This model requires the work output to be specified as well as source and sink heat conditions. Similarly, Little *et al.* [4] model various cycles with a constant heat input using the software package EES. In this study, we aim to optimize the work output from the cycle, so an alternative modelling strategy is used in

order to understand the effect of changing heat input into the cycle. Katsonos *et al.* [5] model an ORC with heat supplied by exhaust gas of a diesel engine. Heat input to the ORC is variable because it is modelled as a function of available heat supply, pinch point temperature difference and heat exchanger (HEX) size. This approach means that the potential improvement in brake-specific fuel consumption of an engine is modelled within practical boundaries. A similar modelling strategy is adopted in this study.

Tchanche *et al.* [6] present a review of ORCs by application. The technical differences of ORC systems and the maturity of existing ORCs are assessed for each application. While this approach gives a thorough understanding of each case, it is not clear whether a generalized approach to optimal ORC design is possible. Similarly, organic cycles using a range of working fluids are reviewed by Chen *et al.* [7]. Fluids are grouped according to how dry or wet they are and an effort is made to determine the most appropriate cycle configuration for each group. However, as is acknowledged in this study, cycle configuration is strongly dependent on the temperature profile of the heat source, and no general principles were drawn from the analysis.

In the present study, Matlab and FluidProp [8] have been used to build an ORC model capable of simulating different organic fluids over a wide range of operating conditions for any heat source and sink. The model requires input conditions for both the heat source and sink and a pinch point temperature difference for the heat exchange processes into and out of the cycle. An advantage of the model is that the heat input is not fixed, as it is in some other studies, e.g. [2], but determined by pinch point analysis. This means that the maximum heat input to the

system is calculated for each ORC configuration and is therefore correctly modelled as a function of the overall system operation. This is particularly important in waste heat applications, where the source of energy is free and therefore, optimizing the cycle for thermal efficiency is not necessarily the best strategy.

In previous studies [2–5], only one set of heat source and sink conditions are used to investigate ORC behaviour. This study models ORCs powered by three heat sources over a range of energy scales and compares results of each case. The heat sources are energy recovery from exhaust heat of large diesel internal combustion (IC) engines, geothermal brines as a by-product of oil and gas production and waste steam from a typical process industry plant. The aim of this study is to further understanding of the influence of the heat source temperature profile on ORC design, and to identify general design principles for ORCs over a range of energy scales.

The heat sources modelled are described in the next section. The ORC model used to analyse the case studies is described in Section 3, before the results from the model are discussed and compared in Section 4. Finally, conclusions are drawn from the work concerning to what extent a general approach to ORC design is possible.

## 2 CASE STUDIES

### 2.1 I.C. engines

Several ORC waste heat recovery (WHR) systems exploiting the waste heat available from IC engines are currently under development, e.g. the US Department of Energy (DOE) funded Super Truck Programme [9]. The DOE recognized that heavy-duty vehicles consume 20% of the fuel used in the USA and that haulage businesses operate on very tight (1–2%) profit margins. This means that there is a strong economic drive to improve specific fuel consumption by using ORC WHR systems. The Super Truck programme aims at improving the specific fuel consumption of Class 8 trucks by 10%. Such trucks use large diesel engines, which typically reject 31% of the raw energy input from the fuel as heat. Exhaust heat characteristics of a fully loaded heavy-duty diesel engine at 1700 rpm are given in Table 1 (taken from Katsanos *et al.* [5]).

**Table 1.** Characteristics of waste heat sources

Case	Mass flow rate $\dot{m}_s$ (kg/s)	Specific enthalpy $h_{s,H}$ (kJ/kgK) $h_{s,H} = f(T_{s,H}, P_s)$	
		$T_{s,H}$ (°C)	$P_s$ (bar)
Full load diesel engine (exhaust)	0.4945	397.8	1
Ninian oil field (hot brine)	1231.3	102	3
Industrial processing plant (saturated steam)	31.94	140	2.3

### 2.2 Hot brines

Ageing oil wells require water injection in order to maintain the pressure and sweep oil from the pore spaces. North Sea petroleum reservoirs are between 2.5 and 5 km deep, and the oil and water mix produced from the well has a temperature in the range 70–150°C. The quantity of hot water produced can be very large (~1000 kg/s), resulting in high potential for large power production from low-grade heat energy ( $Q$ , kW) recovery systems [10]. Platforms currently generate power for water injection pumps by burning gas produced with the oil. However, in wells with falling production the amount of gas produced becomes insufficient and gas must be imported at vast expense. The ORC technology is seen as a potential way to avoid these costs.

Although the idea of using waste water injection brine is relatively new, ORCs driven by geothermal heat are well researched and are in commercial operation, as summarized by Tchanche *et al.* [6]. Nguyen *et al.* [11] studied a typical North Sea oil and gas platform. Exergy analysis was used to identify the most energetically inefficient processes on the platform. The power generation process, by gas turbines, was found to be the least efficient process. Nyugen *et al.* recommend improving thermal efficiency by using ORCs powered by the waste heat from the existing gas turbines. This study models the potential power that an ORC system could produce when supplied with heat from hot brines. An example for the supply of hot brine is taken from the Ninain oil field (see Table 1).

### 2.3 Industrial plant

Many industrial processes produce waste heat, for example the manufacture of cement, textiles and electricity production. The example used in this study is a typical plant supplying steam and electricity to surrounding large chemical processing units. In this plant, steam from a gas and a biomass fuelled boiler is used to produce thermal and electrical energy. Several streams of waste heat are available in the case study used, including flue gas, low pressure and intermediate pressure steam. The potential power production from an ORC fed by a single stream of low pressure waste steam is explored. The characteristics of the low pressure steam flow are included in Table 1.

## 3 RANKINE CYCLE MODEL

A model has been developed to evaluate ORC performance for any waste heat source and heat sink. A schematic diagram of a standard Rankine cycle is shown in Figure 1a. The model is capable of simulating organic cycles of any type, ranging from Trilateral Flash Cycles (TFCs) through to subcritical cycles operating with superheated vapour throughout the expansion in the turbine (Figure 1b). The model has been designed for maximum flexibility. Any heat source and sink conditions can be specified and any ORC cycle can be simulated within user prescribed operating bounds. Standard thermodynamic heat-balance relations

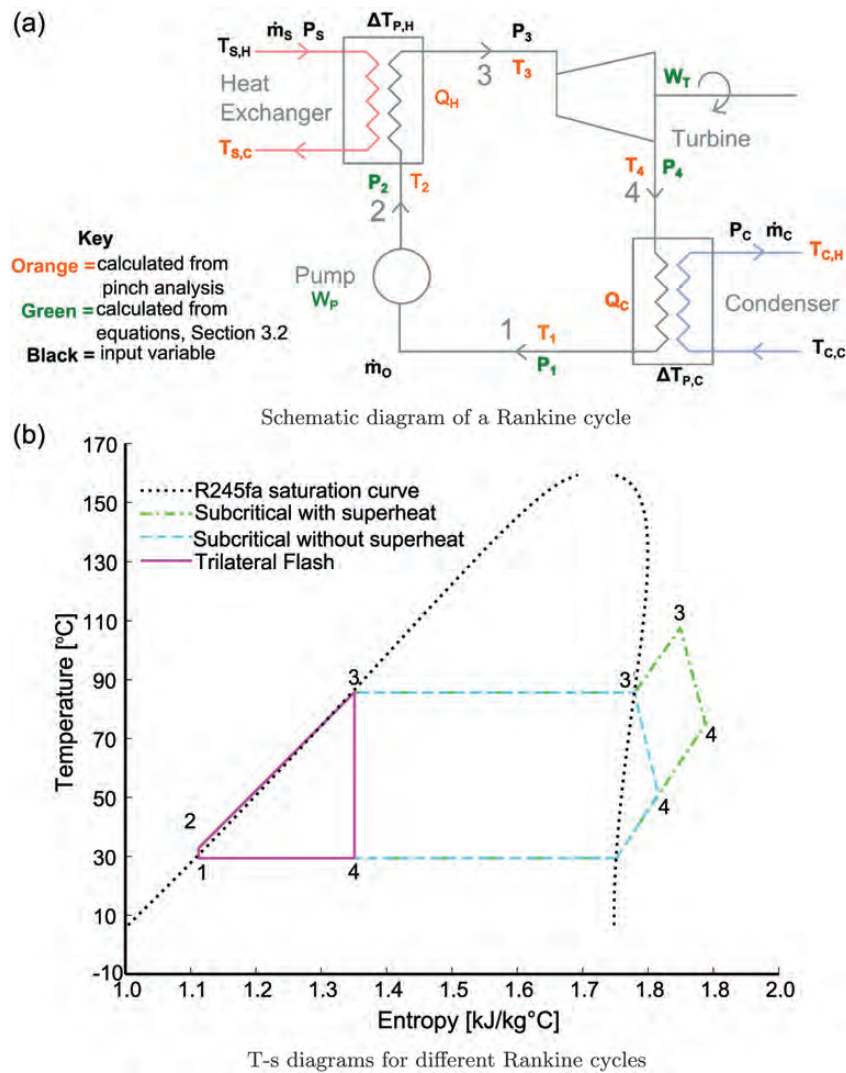


Figure 1. Diagrams of the Rankine cycles modelled in this study. (a) Schematic diagram of a Rankine cycle. (b) T-s diagrams for different Rankine cycles.

are used to describe the fluid pump, heat exchanger, turbine and condenser. FluidProp is used to calculate the thermodynamic properties of the working fluid used in the cycle.

For given heat source and heat sink streams, the mass flow rate and turbine inlet pressure (TIP) of the ORC must be specified in order to model the cycle. Preliminary investigations suggested that the ability of an ORC to recover useful work is less sensitive to the mass flow rate of working fluid around the cycle than it is to TIP. So, the mass flow rate of the working fluid ( $\dot{m}_o$ , kg/s) was fixed at 60% of the waste heat source mass flow rate, in all of the cycle predictions included in this paper. The method of modelling the heat exchange process also assumes that the isobars on the temperature–enthalpy ( $T$ – $H$ ) diagrams in the unsaturated state are straight lines. An upper limit for TIP of  $0.81P_{crit}$ , where  $P_{crit}$  is the property at critical point, was set in order to limit the error associated with this assumption. A lower limit of 1% of the critical pressure was also selected to ensure that the condenser vacuum always remained within practically

achievable limits. Therefore, cycle calculations were only carried out for TIP values in the range of  $0.01P_{crit} < TIP < 0.81P_{crit}$ .

The ORC modelling strategy is shown by the flow diagram in Figure 2, and it is described in detail in the following Sections 3.1 and 3.2.

### 3.1 Heat input, condenser and heat exchanger models

The ORC model has been designed to allow cycle optimization within the boundaries defined by the available operating conditions. The waste heat source and heat sink enthalpy fluxes are needed as inputs for the cycle optimization calculations (see Figure 2). A typical  $T$ – $H$  pinch point diagram is shown in Figure 3 for the ORC heat input process. The enthalpy flux available from the heat source (defined by input variables  $\dot{m}_s$ ,  $P_s$  and  $T_{s,H}$ , see Table 1) and the heat exchanger pinch point temperature ( $\Delta T_{P,H}$ ) are input parameters.



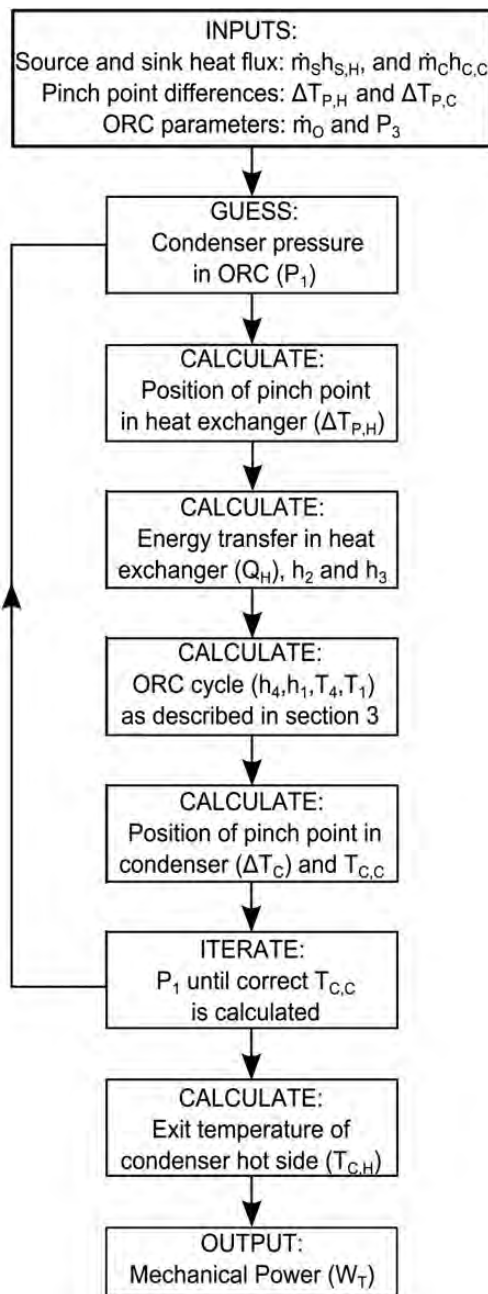


Figure 2. Flow diagram of the ORC model.

A pinch point analysis is carried out using straight line approximations of both source heat and working fluid  $T-H$  profiles. The temperature difference between the maximum cycle temperature possible,  $T_{3,\max} = T_{S,H} - \Delta T_{P,H}$ , and the temperature at the liquid saturation point ( $T_{\text{sat}}$ ) divided by the enthalpy change of the working fluid between these points defines a gradient ( $\phi$ ), calculated by

$$\phi = \frac{T_{3,\max} - T_{\text{sat}}}{\dot{m}_O(h_{3,\max} - h_{\text{liquid,sat}})}, \quad (1)$$

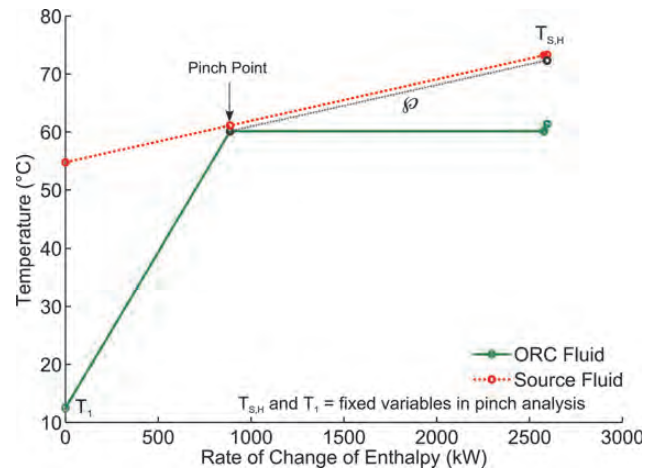


Figure 3. Example  $T-H$  profiles of the heat source and ORC fluid. ORC fluid is R134a at TIP = 16.95 bar and  $\dot{m}_O = 11.99$  kg/s. Heat source fluid is water,  $\dot{m}_S = 33.39$  kg/s and  $T_{S,H} = 73.33$ .

This gradient determines the location of the pinch point. If the  $T-H$  gradient of the heat source fluid,  $dT/dH$ , is shallower than  $\phi$ , then the pinch point must be at the hot end of the heat exchanger. If the  $T-H$  profile of the source heat is steeper than  $\phi$ , the location of the pinch point must be at the cold end or at the liquid saturation point of the working fluid. The precise location of the pinch point can easily be determined by comparing the two gradients.

The pinch point analysis allows the specific enthalpy ( $h$ , kJ/kg) of the ORC fluid at inlet and exit to the heat exchangers,  $h_2$  and  $h_3$ , to be determined (see Points 2 and 3 in Figure 1).  $\dot{m}_O$  was set to 60% of  $\dot{m}_S$  in all calculations as discussed previously. Once  $h_2$  and  $h_3$  are known, the heat input to the ORC cycle,  $Q_H$ , and the enthalpy of the heat source fluid at exit from the heat exchanger,  $h_{S,C}$ , can be determined from the steady flow energy equation (SFEE, Equation 2),

$$Q_H = \dot{m}_O(h_3 - h_2) = \dot{m}_S(h_{S,H} - h_{S,C}), \quad (2)$$

where  $h_{S,H}$  and  $h_{S,C}$  are the specific enthalpies of source, heat and cold sides, respectively.

An identical pinch point analysis is used to calculate the condenser heat exchange process in the model. The enthalpy flux of the heat sink cooling flow is calculated from the input parameters:  $\dot{m}_C$ ,  $P_C$  and  $T_{C,C}$ , where C is the condenser. The condenser pinch point temperature difference ( $\Delta T_{P,C}$ ) is sufficient to be able to determine the enthalpy of the working fluid at condenser exit ( $h_1$ ) for any value of working fluid enthalpy at the condenser inlet ( $h_4$ ), from the SFEE. The fluid in the cool side of the condenser is water in the calculations presented in this paper. The inlet temperature of the cooling water was fixed at 8°C, and its mass flow rate was fixed at  $5\dot{m}_O$  in all cases. The cooling water inlet temperature is representative of a cold water supply in the UK.

The temperature changes of the fluids over the heat exchangers are not fixed. Rather, the pinch point analysis determines the maximum temperature drop and, thus, the amount of heat transfer possible for the specified pinch point temperature differences.

### 3.2 ORC cycle

The ORC model uses standard thermodynamic relations to calculate the fluid conditions at each point in the cycle indicated in Figure 1b. The isentropic efficiency of the turbine and the pump was fixed at 0.8 and 0.75, respectively, for the calculations described in this paper.

Feed pump work ( $W_p$ ) is calculated from the pressure rise across the pump,  $\Delta P$ , where  $P$  is the pressure, and is related to the specific enthalpy increase of the working fluid passing through the pump, by the following equation:

$$W_p = \dot{m}_O \frac{\Delta P}{\eta_p \rho} = \dot{m}_O (h_2 - h_1). \quad (3)$$

The work output from the turbine ( $W_T$ ) is related to the isentropic (I) enthalpy rise across the turbine according to

$$W_T = \eta_T \dot{m}_O (h_3 - h_{4,I}), \quad (4)$$

where  $\eta$  is the efficiency.

Turbine outlet specific enthalpy,  $h_4$ , is calculated from

$$W_T = \dot{m}_O (h_3 - h_4) \Rightarrow h_4 = h_3 - \left( \frac{W_T}{\dot{m}_O} \right). \quad (5)$$

This is limited in the model to a minimum saturated vapour state corresponding to a dryness fraction of 0.75. At lower vapour qualities, the vapour is considered too wet to ensure the mechanical integrity of the turbine. Turbine exit vapour quality

is therefore one of the parameters that defines the design space within which meaningful cycle simulations can be carried out using the model.

### 3.3 Fluids modelled

The fluids modelled are dry or isentropic, as recommended by Tchanche *et al.* [6]. Dry fluids have a positive vapour saturation curve, whereas isentropic fluids have a vertical saturation curve. This results in the fluid remaining in a superheated vapour state as it expands through the turbine in an ORC. Therefore, ORCs do not need high levels of superheating in order to avoid excessive wetness at turbine exit, unlike water/steam cycles. The fluids selected for the current ORC calculations are R245fa, *n*-pentane, *n*-octane, toluene, cyclohexane and R134a (isentropic). As the three heat sources considered have a wide range of temperatures, working fluids with a range of critical temperatures were selected to ensure ORCs for each source were possible within the defined operating limits and operating conditions of the fluids. Saturation curves for all fluids used are shown, a  $T$ - $S$  diagram, together with water, in Figure 4.

### 3.4 Model validation

Aneke *et al.* [3] compare the performance of a model built with IPSEpro, using REFPROP 7.0 to calculate fluid properties, with real data from the Chena binary geothermal power plant. The cycle parameters used in the simulations run in Aneke *et al.*'s study have been input into the model used in this study. The results of both the simulations are shown in Table 2.

The results from both models agree closely. The net power output and overall cycle thermal efficiency calculated by the two

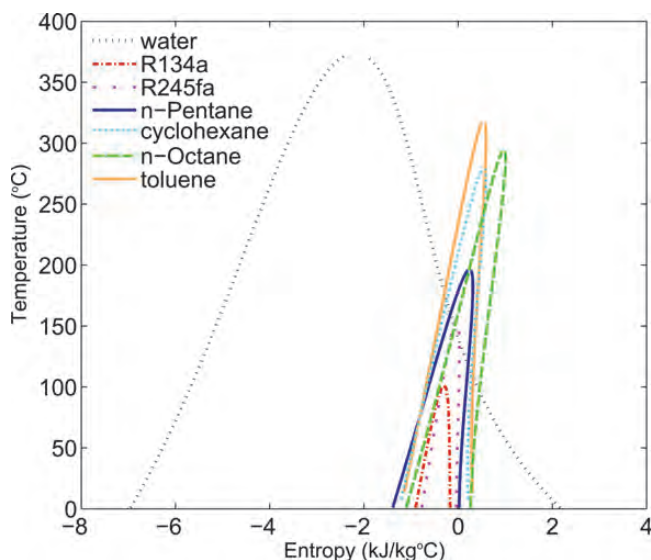


Figure 4.  $T-S$  plots for water and the selected organic working fluids.

Table 2. Comparison of results from the model by Aneke *et al.* [3] and the present model

Parameter	Aneke <i>et al.</i> model	Present model	Relative error (%)
Geothermal fluid mass flow rate (kg/s)	33.39 <sup>a</sup>	33.39 <sup>a</sup>	—
Geothermal fluid temperature (°C)	73.33 <sup>a</sup>	73.33 <sup>a</sup>	—
Cooling water mass flowrate (kg/s)	101.68 <sup>a</sup>	101.68 <sup>a</sup>	—
Cooling water source temperature (°C)	4.44 <sup>a</sup>	4.44 <sup>a</sup>	—
Working fluid	R134a <sup>a</sup>	R134a <sup>a</sup>	—
Turbine efficiency	0.8 <sup>a</sup>	0.8 <sup>a</sup>	—
Turbine inlet pressure (bar)	16.95 <sup>a</sup>	16.95 <sup>a</sup>	—
Turbine outlet pressure (bar)	4.39 <sup>a</sup>	4.41	0.39
Gross generator power (kW)	250 <sup>a</sup>	254.81	1.69
Pump power (kW)	40 <sup>a</sup>	39.97	−0.08
Pump efficiency	0.305 <sup>a</sup> <sub>inferred</sub>	0.305 <sup>a</sup>	—
Geothermal exit temperature (°C)	54.94	54.76	−0.32
Cooling water exit temperature (°C)	9.91	9.92	0.10
Working fluid mass flow rate (kg/s)	11.99	11.99 <sup>a</sup>	—
Net plant power (kW)	210	214.25	2.03
Thermal efficiency	0.08	0.083	3.22
Evaporator heat transfer rate (kWth)	2570.38	2594.63	0.94
Condenser heat transfer rate (kWth)	2327.1	2342.01	0.64
Pinch point temperature difference (°C)	—	1 <sup>a</sup>	—

<sup>a</sup>Indicates input variable

methods differ by 1.69 and 3.22%, respectively. This good agreement is despite the very different modelling approaches used in the two methods. The current method uses pinch point analysis as described earlier, whereas Aneke *et al.*'s method relies on prescribing the power output from the ORC to fully define the cycle. This benchmarking exercise against Aneke *et al.*'s work validates the method used in the present study.

## 4 MODEL RESULTS AND DISCUSSION

Each case presented in Table 1 was modelled with each of the fluids described in Section 3.3. Figure 5 shows thermal efficiency, heat input and mechanical work against TIP for geothermal brine produced by the Ninian oil field. Thermal efficiency of the

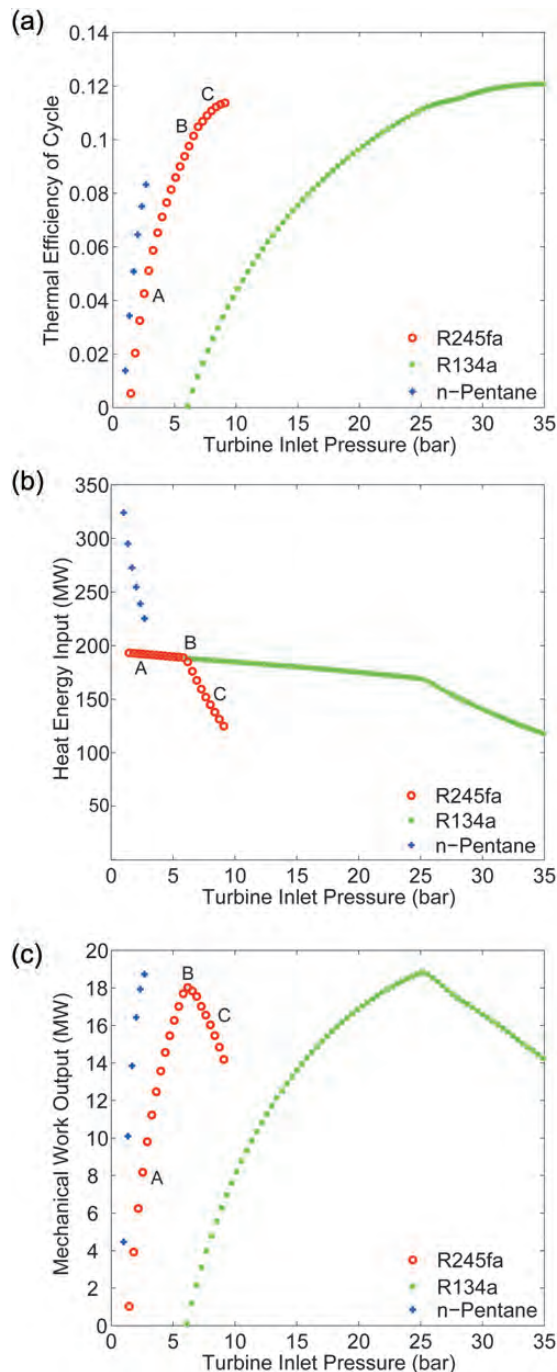


Figure 5. Plots of efficiency (a), heat input (b) and mechanical work output (c) against TIP for the Ninian hot brine case using R245fa as the working fluid (Table 1).

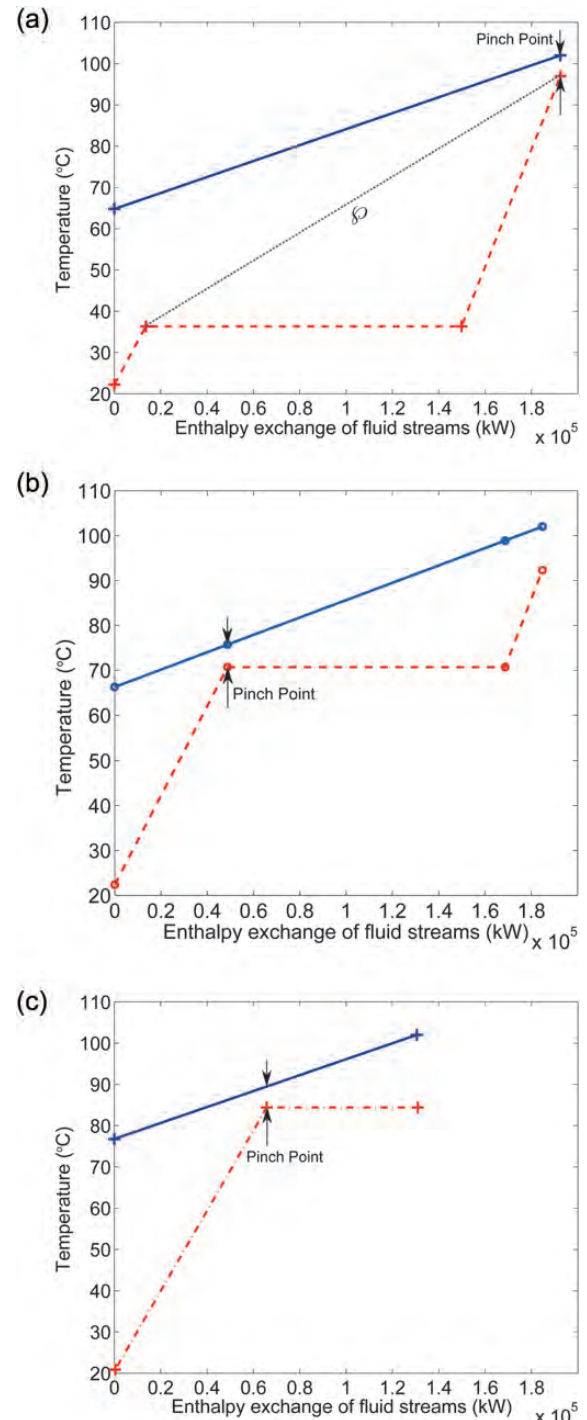


Figure 6.  $T - H$  plots at the heat exchanger for cycles A, B, C using R245fa as the ORC working fluid, as labelled in Figure 5.



ORC increases with TIP but the maximum mechanical work output is achieved at  $P_3 = 25.59$  bar for R134a and at 6.19 bar with R245fa as the working fluid. The range of TIP for which the calculations were performed is limited by the physical limits of the working fluids imposed by REFPROP and boundaries imposed in the model as described in Section 3. In the case of the hot brines, only results calculated for R134a, R245fa and *n*-pentane fall within these limits, and are presented in Figure 5.

The existence of a maximum work output relates to the amount of heat that can be input into the cycle as the operating conditions vary. The Aneke *et al.*'s study also shows similar trends [3]. Figure 6 shows plots of the temperature of the two fluid streams as they flow along the heat exchanger (this distance is expressed in terms of the enthalpy exchange between the fluid streams in the figure), for the Ninian oil field hot brine case with R245fa. The three plots are at the different locations labelled in Figure 5. These are below the maximum work output (Cycle A, Figure 6a), at the maximum (Cycle B, Figure 6b) and above the maximum (Cycle C, Figure 6c). The plots show how heat input decreases as TIP increases, as seen in Figure 5b. As TIP increases, from Cycles A–C, a smaller temperature drop occurs across the heat source stream in Figure 6, so less heat energy is transferred into the ORC working fluid.

Figure 7 shows the temperature entropy plots for Cycles A, B and C. Cycle C is the most thermally efficient, and Cycle A is the least thermally efficient. A larger amount of work is produced by Cycle B than by Cycle A, because the increase in efficiency outweighs the decrease in heat energy into the cycle. After Point B heat input to the cycle decreases more rapidly as TIP increases, due to the change in pinch point location. So for Cycle C, the gain in efficiency of the cycle is outweighed by the reduction in heat into the cycle and less work output is produced. These figures demonstrate that an optimum balance between heat

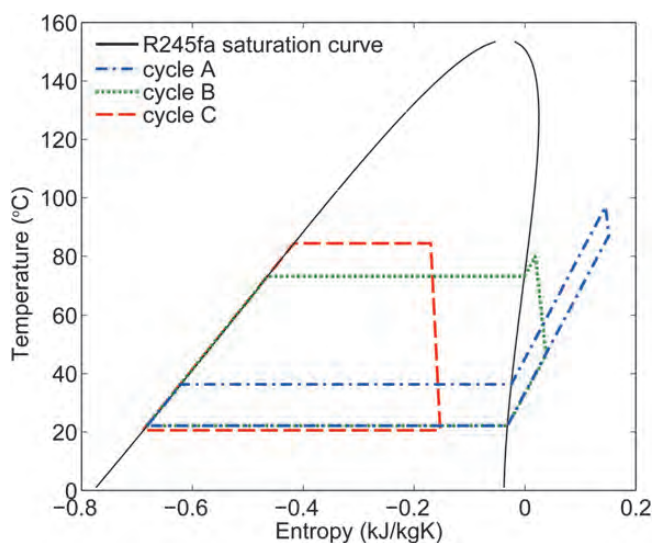


Figure 7. Temperature entropy diagrams for cycles A, B and C labelled in Figure 5.

input into the cycle and thermal efficiency, of the cycle must be reached in order to maximize the work output.

Work output for each waste heat source can be compared by considering Figures 5c, 8b and 10b. Some working fluids modelled in the hot brine and industrial waste steam cases show a curve that has an optimum TIP, where work output is maximum. But others, for example the diesel engine results (Figure 10b), show an increase in work output with TIP for all working fluids simulated, in the case of the fully loaded engine. In all cases, the ORC configuration varies from subcritical or tri-lateral to superheated, but the optimum cycle configuration, with respect to work output, is not common in all cases. The trend of these plots can be explained by examining the location of the pinch point and the effect this has on the heat input to the cycle, in each case.

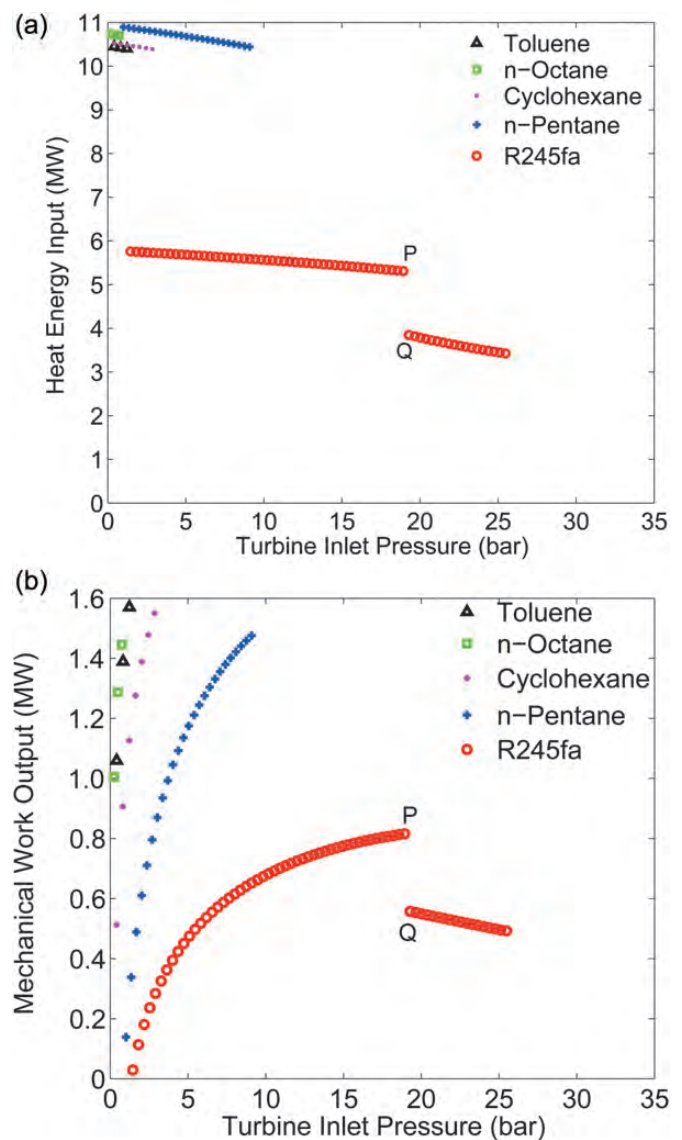


Figure 8. Plots of heat input (a) and mechanical work output (b) against TIP for the industrial steam case (Table 1).



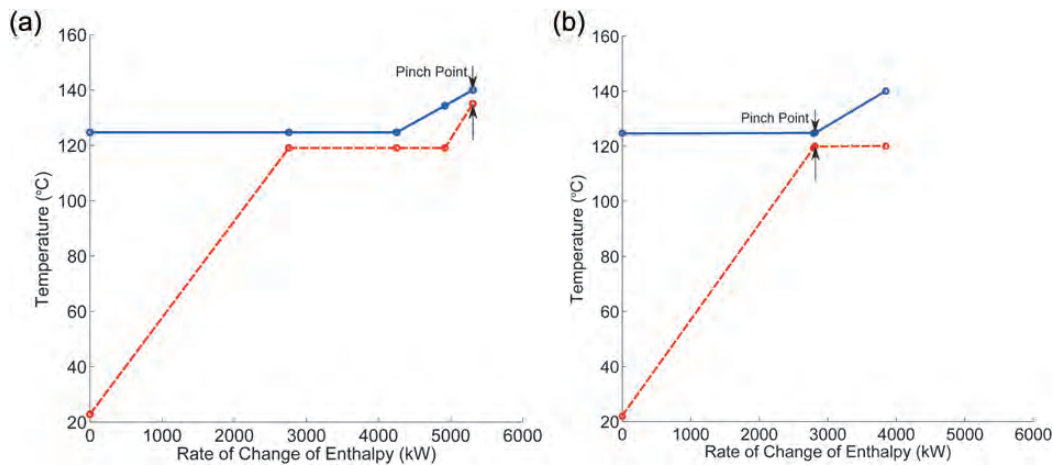


Figure 9.  $T-H$  plots of the heat exchange process for the industrial steam case, as labelled in Figure 8.

The pinch analysis in each case uses the  $T-H$  gradient of the heat source fluid compared with the gradient of the ORC working fluid,  $\phi$  (see Equation 1), to determine the location of the pinch point (see Section 3.1). In the case of the hot oil field brines with R245fa as the ORC working fluid, the gradient of the source heat fluid is less steep than the ORC working fluid gradient, as seen from Figure 6a. This results in the pinch point being initially located at the highest temperature point of the ORC working fluid (i.e. heat exchanger exit/turbine inlet). The pinch point moves to the working fluid liquid saturation point as the TIP increases to 6.19 bar, Figure 6b. The change in the location of the pinch point increases the rate at which the heat input to the cycle decreases with TIP, as seen in Figure 5b.

Similarly, in the case of industrial steam with R245fa ORC working fluid there is an optimum TIP for maximum work output after which the work output decreases, that is 18.93 bar as shown in Figure 8b. Figure 8a shows that the rate of heat input into the cycle decreases at the optimum TIP, 18.93 bar, in the same way as for the Ninian hot brines results (see Figure 5b). However, there is a significant discontinuity in both the plots of heat input and work output against TIP at the optimum TIP (Figure 8). This is not seen in the Ninian hot brine results. The change in the gradient of work output with TIP is due to the change in location of the pinch point in the same way as previously described for the hot brines case. The discontinuity is a result of the phase change of the heat source fluid. The phase change means that when the location of the pinch point moves from the hot end of the heat exchanger (Cycle B, Figure 9a) to the liquid saturation point of the ORC working fluid (Cycle Q, Figure 9b) much less heat can be taken into the cycle. This is because for Cycle Q, the heat input to the cycle is limited by the enthalpy change between the pinch point and hot temperature of the working fluid, which is quite small. Cycle Q is close to a trilateral flash cycle and so may not be possible to implement practically. However, for this test case whether Cycle Q is practical or not, these results show that it would be extremely detrimental to cycle performance if the TIP is taken above 18.93 bar with R245fa.

The remaining cases show an increase in work output with TIP. These cases are firstly, all the ORC working fluids simulated in the diesel engine case (Figure 10b). Secondly, the ORC working fluids toluene, *n*-octane, cyclohexane and *n*-pentane in the industrial steam case (Figure 8b) and finally, *n*-pentane in the hot brines case (Figure 5c). As previously shown, the change in pinch point location can cause the rate of change of heat input into the cycle to decrease more quickly with TIP, causing work output of the cycle to also decrease. For these cases, the rate at which the heat input to the cycle decreases with TIP does not change significantly, as shown in Figures 5b, 8a and 10a. This means that the change in work output of the cycle with TIP is primarily influenced by thermal efficiency of the cycle. In all cases, the cycle thermal efficiency increases with TIP. Therefore, in cases where the heat input does not change significantly with TIP, the cycle work output will increase with TIP. The results for toluene in the diesel engine case show a small perturbation of heat input to the ORC at low TIPs (Figure 10a). Investigation into this behaviour showed that it was due to the straight line approximation of the liquid saturation curve used in the pinch point analysis. The change of the heat input into the cycle this caused is small, and therefore, is a second-order effect which does not change the trend in work output with respect to TIP (Figure 10b).

The results show that the location of the pinch point changes with TIP. The location of the pinch point strongly influences cycle performance and is dependent on the source heat fluid  $T-H$  profile. The thermal efficiency of the cycle always increases with TIP. But, in cases where the  $T-H$  gradient is shallower than the working fluid isobar, the heat input into the cycle may decrease at a faster rate than the increase in efficiency. This will result in a decrease in a peak in work output from the cycle at some optimum TIP.

Overall, Figures 5c, 8b and 10b show that a similar maximum work output is achieved with several different ORC fluids in each case. The exceptions are *n*-pentane in the engine and hot brine case and R245fa in the industrial steam case. These results

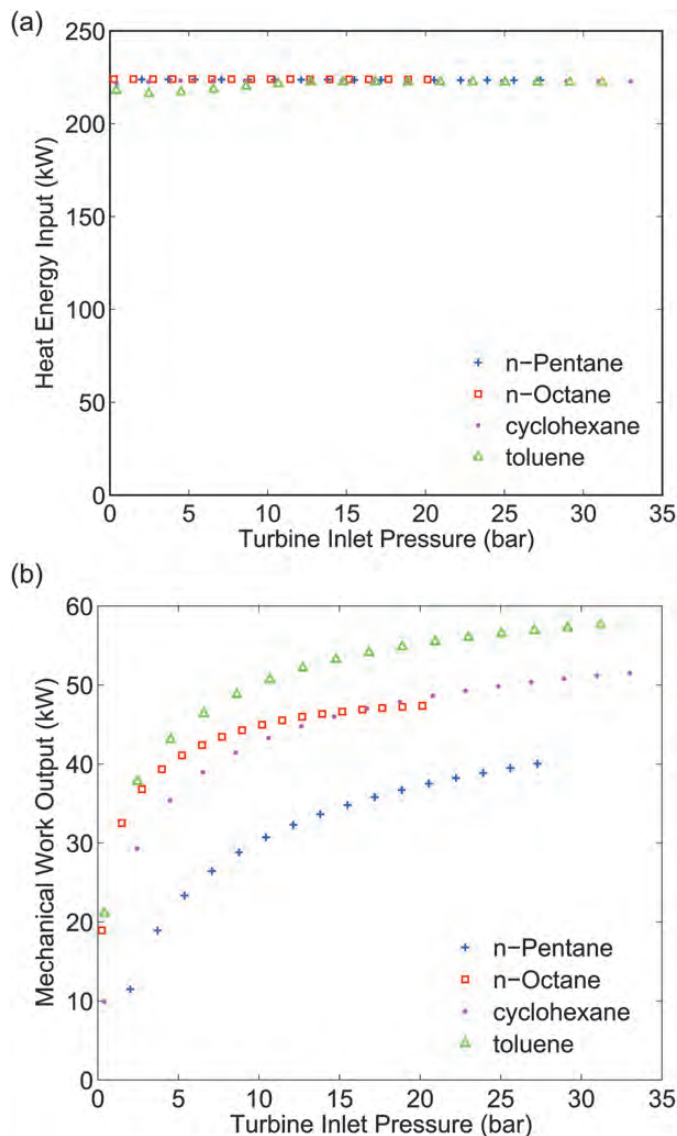


Figure 10. Plots of heat input (a) and mechanical work output (b) against TIP for the diesel engine case (Table 1).

indicate that, under the constraint imposed in this work, i.e.  $\dot{m}_O = 0.6 \dot{m}_S$ , the choice of ORC working fluid does not strongly influence the cycles ability to recover work. Although, the choice of working fluid does influence the TIP at which maximum energy recovery is achieved.

## 5 CONCLUSIONS AND FURTHER WORK

It is desirable to optimize mechanical work output (not cycle efficiency) when designing ORC systems to recover useful power from waste heat. This study considered three potential sources of waste heat. ORCs were modelled for these sources with the aim of further understanding the influence of the  $T-H$  profile of the heat source fluid on the optimal cycle configuration, and to

identify general design principles for ORCs over a range of energy scales. The ORCs were modelled with a fixed mass flow rate and use the available heat source and sink conditions to compute possible ORCs based on the fixed pinch point temperature differences in the cycle heat exchangers. Simulations were carried out for each heat source with the following working fluids: R245fa, *n*-pentane, *n*-octane, toluene, cyclohexane and R134a.

It has been shown that the optimal cycle configuration (sub-critical, superheated or TFC) for recovered work output is strongly dependent on the  $T-H$  profile of the heat source fluid. General design principles can be drawn from comparing the gradient of the ORC working fluid, as defined by Equation 1, with the gradient of the heat source fluid  $T-H$  profile. Where the gradient of the source heat fluid  $T-H$  profile is steeper than that of the working fluid, cycle work output will increase with an increase in TIP. However, care should be taken when the gradient of the source  $T-H$  profile is shallower than that of the ORC working fluid, under these conditions too large a TIP can result in a decrease in work output from the cycle.

The results show that the conditions for optimizing mechanical work output from an ORC are different from those for optimum thermal efficiency of the cycle. The maximum work output is achieved under conditions where there is the best balance between the heat input to the cycle (decreases with TIP) and thermal efficiency (increases with TIP). ORC cycles powered by waste heat should be designed to operate in the region of the maximum work output, if possible, but avoiding conditions that might cause the pinch point to change and therefore the rate heat taken into the cycle to decrease. This is particularly significant for cases where the source heat fluid changes the state, for these heat sources change in pinch point can cause a sudden step decrease in work output from the cycle.

The results indicate that, for a fixed ORC working fluid mass flow rate, fluid selection does not have a strong influence on the ORCs ability to recover waste heat. But ORC working fluid choice does influence the TIP at which the maximum work output is achieved.

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